

# Estimation of the elastic properties of an adhesive joint of thermoplastic honeycomb sandwich panels through analysis of structural resonant behaviour

**S. Debruyne<sup>1</sup>, D. Vandepitte<sup>2</sup>, K. Van Massenhove<sup>1</sup>**

<sup>1</sup> KU Leuven @ Kulab, Department Mechanical Engineering

Zeedijk 101, B-8400 Oostende, Belgium

e-mail: [stijn.debruyne@kuleuven.be](mailto:stijn.debruyne@kuleuven.be), [korneel.vanmassenhove@kuleuven.be](mailto:korneel.vanmassenhove@kuleuven.be)

<sup>2</sup> KU Leuven, Department Mechanical Engineering

Celestijnenlaan 300 B, B-3001, Heverlee, Belgium

e-mail: [dirk.vandepitte@kuleuven.be](mailto:dirk.vandepitte@kuleuven.be)

## Abstract

In recent years the use of adhesive bonds for structural joints has increased greatly. Currently, many different material types are combined in a sole product. Especially in applications where lightweight materials are used, adhesive bonded joints offer distinct advantages over traditional joining techniques like welding, bolting and riveting. Important features are the possibility to join different types of materials and the ability to smoothen structural stress concentrations.

A lot of research work has already been performed in the field of adhesives. However, it mainly focuses on the more fundamental physical-chemical properties of the different adhesive types, substrate adhesion and theoretical models that describe the viscoelastic behaviour of adhesives.

There is a high number of environmental parameters that influence the elastic behaviour of an adhesive joint. Therefore, also the vibration behaviour of adhesively bonded structures is a process that is subjected to many uncertainties.

This paper focusses on the joining of glass fibre reinforced polypropylene honeycomb sandwich panels by means of an adhesive bond. The simple case of joining two rectangular panels is considered.

A first part discusses the adhesive type and bonding application. It thereby focusses on the estimation of elastic properties of the bonded zone. The uncertainty involved is quantified where possible.

The second part estimates the impact of the uncertainty on bonding parameters on the resonant behaviour of the two joined honeycomb panels. As reference data, experimentally determined resonance frequencies and mode shapes under free boundary conditions are used. This part fully discusses how finite element modelling is used to estimate the bonding layer's shear modulus.

The third part discusses the influence of vibration frequency, temperature and amplitude on the bonding layer's shear modulus. Various validation experiments are considered.

Finally, the paper summarizes the research work and prospects to further application driven research.

## 1 Introduction

Honeycomb sandwich panels are often part of large light weight structures, e.g. truck bodies. Therefore the individual panels have to be joined. Next to the traditional techniques, such as riveting, adhesive bonding is one way of connecting individual panels. This research focusses on joining thermoplastic honeycomb sandwich panels, made from polypropylene. Due to its physical nature this material is not ideally suited for adhesive bonding. Joining polypropylene structures by means of a welding process is the most effective. A good alternative to this is adhesive bonding using a hot-melt glue connection.

In the past decades a lot of research work has been carried out on adhesive joints [1]. On the one hand, many publications focus on analytical models that predict the stress – strain relations of different glue types, including viscoelastic ones. On the other hand much research work is performed on the finite element modelling of adhesively bonded joints [2 - 6]. Up till now, most research efforts focus on strength and failure modelling of adhesively bonded joints [7 – 9]. The study of failure mechanisms is most often performed in the context of adhesively bonded composite materials. Non – ductile epoxy glues are used there.

Grant [12] and Da Silva [13] describe the analysis and modelling of more ductile bonded joints, suited for the automotive industry. They study the effect of temperature on the mechanical performance of the joint and general modelling strategies of the adhesive joint.

However, very little work has been published on the characterization of the elastic properties [14 - 17] of the bonding zone, its effect on the dynamic properties of a whole structure and the uncertainty assessment involved. This paper therefore focusses on the estimation of the elastic properties of the bonding zone of two joined thermoplastic honeycomb sandwich panels by means of experimental and numerical modal analysis.

A first part of this paper specifies the test sample used in this research. It describes the main properties of the individual panels and outlines the properties of the bonded joint.

The second part of the paper illustrates the process of experimental modal analysis to obtain natural frequencies and corresponding mode shapes through hammer excitation. Experiments are carried out in ambient (20°C) and elevated (35°C) temperatures.

The third part focusses on the numerical modal analysis using a finite element model of the two joined rectangular panels. It describes the estimation of initial elastic material properties of the glue and it illustrates the modelling principles used [18 - 21]. The finite element model calculates natural frequencies and mode shapes under free boundary conditions.

The fourth and most extensive part discusses the process of estimating the elastic properties of the glue material as a function of vibration frequency and temperature. This is done by updating the finite element models [22] with respect to the experimentally determined natural frequencies and mode shapes. The model parameters of interest are the shear and Young's modulus of the assumed isotropic glue material.

The final chapter summarizes with some main conclusions and prospects to further research activities in this area.

## **2 Test samples**

This research work makes use of commercially available MonoPan® panels. These are honeycomb sandwich panels that consist of a cylindrical polypropylene honeycomb core and glass fibre weave

reinforced skin faces. Table 1 gives an overview of the main parameters of the two nominally identical panels that are used.

Length (mm)	594
Width (mm)	420
Panel thickness (mm)	15
Skin thickness (mm)	1.3
Areal mass (kg/m <sup>2</sup> )	5.4

**Table 1:** Dimensional properties of MonoPan® sandwich panels.

The two rectangular panels are joined by means of two aluminum strips with a width of 60 mm and a thickness of 1.35 mm. By its nature, polypropylene structures are difficult to bond with any adhesive. Thermal welding of this thermoplastic material is by far the most efficient approach. However, the use of a hot-melt adhesive produces a hybrid between adhesive bonding and welding. This research uses Macromelt Q5375 (Loctite). Its basic component is polypropylene. By means of a heat gun the glue can be applied to the bonding surfaces; it is heated up to 180°C. To increase adhesion, the aluminum strips are oven heated to 80°C. Figure 1 illustrates the two joined panels.



**Figure 1:** Two joined thermoplastic honeycomb panels.

The nominal thickness of the adhesive layer is 1.33 mm.

### 3 Experimental modal analysis

The joined honeycomb panels are subjected to a vibration analysis. The bonded structure is thereby elastically suspended and excited with an impulse hammer. Table 2 gives an overview of the obtained natural frequencies when conducting the analysis at 20 °C and 35 °C. For comparison,

	Mode	1	2	3	4	5	6	7	8
Joined panels	Frequency @ 20 °C	41.5	60.5	94.5	146.5	191.5	226	323	557
	Frequency @ 35 °C	38.5	54.5	88	143	184	215	316	523
	Relative difference (%)	-7.2	-9.9	-6.9	-2.4	-3.9	-5.7	-2.2	-6.1
Single panel	Frequency @ 20 °C	79.5	134	207.5	279.5	318	345	395	433.5
	Frequency @ 35 °C	77.8	130	205	278	314	338	386	427
	Relative difference (%)	-2.1	-3	-1.2	-0.54	-1.3	-2	-2.3	-1.5

**Table 2:** Experimentally obtained natural frequencies for the first eight modes.

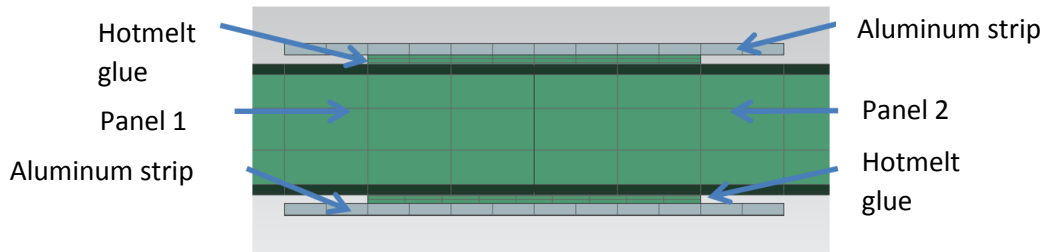
Table 2 clearly shows that the structures become less stiff at elevated temperatures. The relative frequency decrease in case of the adhesively joined panels is higher than in case of a single panel. This is an indication that the stiffness of the adhesive layer decreases stronger than the honeycomb panel stiffness

parameters. Since the deformation at the bonded zone varies from mode to mode, it is expected that mode shapes characterized by a strong shear deformation of the bonded zone show a large frequency decrease. This is evident when observing modes 1 (torsional) and 2 (bending).

## 4 Numerical modal analysis

### 4.1 Modelling principle

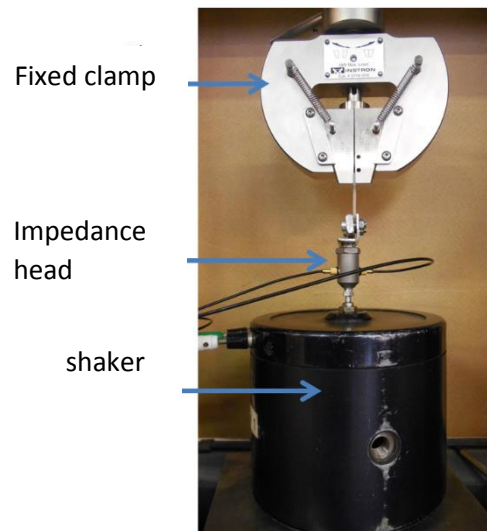
The estimation of elastic material properties from structural resonant behaviour is an inverse problem. For reasons of computational efficiency the finite element models used, have to be simple but adequate. This research uses the shell-volume-shell approach [22] to model the honeycomb sandwich panels. This approach enables the bonding layer to be modelled with volume elements and the aluminum strips as shell structures. Figure 2 illustrates this modelling principle.



**Figure 2:** shell-volume-shell modelling of two adhesively bonded honeycomb sandwich panels.

The glue is assumed to behave isotropically. The core and skin faces of the honeycomb panels are modelled as orthotropic materials following [22].

A series of tensile tests is carried out to estimate the Young's modulus of the hot-melt glue. Herefore, tensile test specimens of pure glue are fabricated. At low strain rates up to 5 % the material behaves linearly. Fifteen tests are conducted; this yields a Young's modulus of 6.5 MPa and a corresponding coefficient of variation (COV) of 4 %. The test specimens are also subjected to dynamical material analysis (DMA) within a frequency range of 10 to 400 Hz. An electro dynamic shaker acts as a vibration actuator and an impedance head measures the applied force and acceleration, as illustrated by figure 3. The applied static strain during the experiments is 5 % and the experiments are conducted at 20 °C.



**Figure 3:** set-up for dynamical material analysis measurements.

Expression (1) calculates the resulting complex modulus  $E^*$ .

$$E^*(j\omega) = H(j\omega) \frac{L_0}{S(j\omega)^2} \quad (1)$$

In (1), the frequency response function  $H(j\omega)$  is the ratio of the measured force and acceleration signals.

Table 2 gives an overview of the obtained complex Young's moduli at different frequencies from 0 to 150 Hz.

Frequency (Hz)	0	10	30	50	70	90	110	130	150
Complex modulus (MPa)	6	8	10	11	13	14	16	19	21
Loss factor	0	0.09	0.08	0.08	0.09	0.07	0.05	0.04	0.02

**Table 3:** Experimentally determined complex Young's moduli of hot-melt glue.

In this frequency range the loss factor remains constant which indicates that glue material damping does not vary much within the observed frequency range.

## 4.2 Results

Using an initial value of 10 MPa for the Young's modulus and 0.43 for Poisson's ratio, the finite element model is used to calculate the first eight natural frequencies and mode shapes of the joined panels. Table 4 gives an overview and makes the comparison with the experimentally determined counterparts.

Mode	1	2	3	4	5	6	7	8
FE frequency	40.2	59.6	93.8	148.3	195.4	216.8	336.3	584
Experimental frequency @ 20 °C	41.5	60.5	94.5	146.5	191.5	226	323	557
Experimental frequency @ 35 °C	38.5	54.5	88	143	184	215	316	523
Relative difference @ 20 °C (%)	-3.1	-1.5	-0.75	1.2	2	-4.1	4.1	4.8
Relative difference @ 35 °C (%)	4.4	9.4	6.6	3.7	6.2	0.8	6.4	12

**Table 4:** Numerically calculated compared to experimentally determined natural frequencies for the first eight modes.

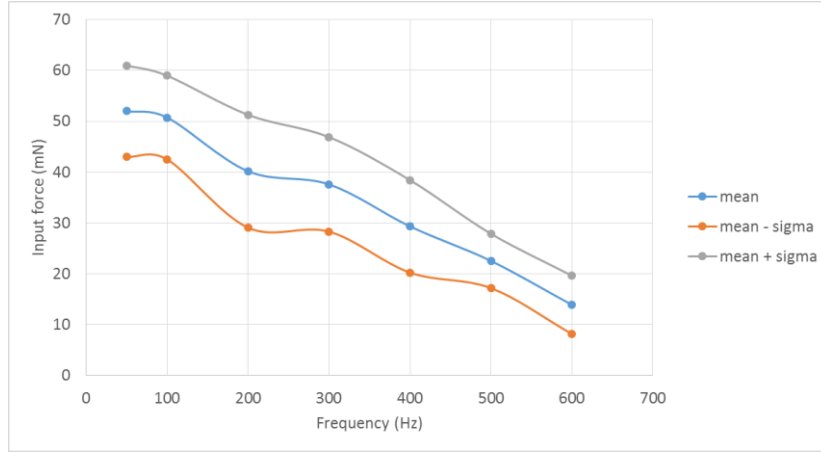
From table 2 it is clear that for a separate panel (not glued) a temperature rise of 15 °C leads to a natural frequency decrease of approximately 2 %. This corresponds to a stiffness reduction of 4 %. The relative differences of the last row in table 4 can thus be reduced by 2 % if the stiffness of the skin (Young's modulus) and core (mainly out-of-plane) shear modulus are reduced by 4 %. The remaining difference is due to a decrease of adhesive stiffness. This work only focusses on the adhesive and not on the parameters of the panels as such.

## 5 Estimating elastic glue properties

This part describes how the main elastic properties of the adhesive layer are estimated using experimental and numerical modal analysis. Special attention is given to the estimation of damping properties of the adhesive layer.

## 5.1 Finite element model update preparation

The goal of this part is to estimate the elastic properties of the adhesive used to join the two honeycomb panels. Due to the nature of the bonded joint and the kind of mode shape deformations, the shear modulus of the adhesive is a key factor. The glue material however, is characterized by elastic properties that are both temperature and frequency dependent. Estimating material properties by means of a set of experimentally determined natural frequencies and mode shapes implies that the modal participation of each mode had to be studied. Since hammer excitation is used during the experimental modal analysis, the excitation force is not equal for each mode. Therefore the input force spectra have to be considered. Figure 4 shows the mean force spectrum, calculated from a set of input spectra of 80 excitation points in total.



**Figure 4:** Mean input force spectrum.

Figure 4 shows that the input force decreases with increasing frequency. This means that the mean excitation force of mode 8 (557 Hz) is approximately 17 mN. Compared to 52 mN for mode 2 (60.5 Hz) this means that the mean modal participation of mode 8 is approximately three times less than that of mode 2. This has to be considered when performing an update of the finite element model. The first mode is subjected to the greatest input force, therefore it is characterized by the reference weighting factor 1 in table 5. The weighting factor of all other seven modes is expressed to this reference.

Mode	1	2	3	4	5	6	7	8
Weighting factor	1	0.99	0.98	0.87	0.77	0.74	0.65	0.33

**Table 5:** Weighting factors for the different modes, needed for finite element model updating

## 5.2 Finite element model updating

The finite element model of the adhesively bonded honeycomb panels is updated with respect to the shear modulus of the adhesive. The Poisson's ratio is kept constant during this procedure.

The factors from table 5 are the individual mode weighting factors in the model updating objective function [22], or often referred to as the cost function. It is expressed by (2).

$$f(\Delta DV_j) = \min \left( \sum_{i=1}^{N_T} A_i |\varepsilon_i| + O \sum_{j=1}^{N_{DV}} B_j |\Delta DV_j| \right) \quad (2)$$

This minimization procedure alters the values of the considered FE model design variables  $DV_j$  (with  $\Delta DV_j$  minimal) to minimise the relative difference between experimental and numerical modal parameters (difference  $|\varepsilon_i|$ ). The latter are natural frequencies and mode shapes (by means of MAC-value of a matching mode pair). The first 8 natural frequencies with corresponding mode shapes are used for FEM updating and are called the ‘updating targets’. Consequently there are 16 updating targets in this research. Both target types have equal weights for the updating process. The design variable weighting factors  $B_j$  and  $O$  (individual and overall) are all set to 1 for each objective function of a design variable  $DV_j$ . However, the individual target weighting factors  $A_i$  are set according table 5.

In this research, the FE model update is performed in eight steps, matching the number of experimental modes used. For each numerical/experimental mode pair the objective function is minimized. The number of design variables  $N_{DV}$  thus equals 1 (only glue shear modulus is variable, Poisson’s ratio is kept constant during the analysis) for each model update while the number of updating targets  $N_T$  equals 2 (natural frequency and MAC – number of the mode pair). A least squares minimization is chosen to minimize the objective function iteratively.

Applying this procedure yields a set of eight values for the adhesive shear modulus. Table 6 gives an overview of the obtained shear moduli at both temperatures of 20 and 35 °C.

Mode	1	2	3	4	5	6	7	8
Experimental frequency @ 20 °C	41.5	60.5	94.5	146.5	191.5	226	323	557
Adhesive shear modulus @ 20 °C	22	20	19	25	29	36	42	50
Experimental frequency @ 35 °C	38.5	54.5	88	143	184	215	316	523
Adhesive shear modulus @ 35 °C	19	18	17	24	26	31	38	44
Relative difference @ 20 °C (%)	-3.1	-1.5	-0.75	1.2	2	-4.1	4.1	4.8
Relative difference @ 35 °C (%)	4.4	9.4	6.6	3.7	6.2	0.8	6.4	12

**Table 6:** Obtained values for the adhesive shear modulus.

### 5.3 Estimation of damping of the adhesive layer

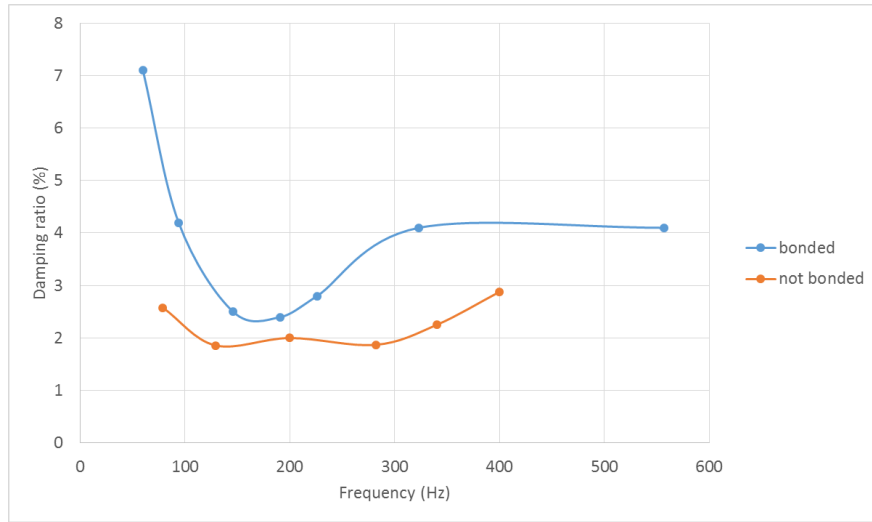
In this case, the applied method enables the estimation of the frequency dependent shear modulus of an adhesive used in a structural bonded joint. The estimated values are complex shear moduli with the ratio of their real and complex parts being the loss factor [3]. Each shear modulus value of table 6 is the modulus of a complex number consisting of a real part which is the storage modulus and an imaginary part which is the loss modulus.

The relation between the loss factor of the adhesive material and the vibration damping of the bonded panel structure is only straightforward in case of pure viscous damping. Expression (3) relates the loss factor  $\eta$  to the structural damping ratio  $\xi$  at resonance.

$$\frac{\omega_2 - \omega_1}{2\omega_n} = \xi = \sqrt{1 + \eta} - \sqrt{1 - \eta} \approx \frac{\eta}{2} \quad (3)$$

According to Ungar and Kerwin [23], the simplification in expression (3) is good within 1% for general damping cases (not only viscous damping) if the loss factor does not exceed 0.28. Taking into account the experimentally determined loss factors of table 3, the simplified expression (3) is thus valid.

In this case, not all vibration damping results from the damping in the adhesive layer but some part is due to the damping in the honeycomb panels. Figure 5 compares the experimentally determined damping ratios of a single honeycomb panel with those of the adhesively bonded example, as a function of frequency.



**Figure 5:** Experimentally determined damping ratios of bonded panel assembly compared with that of a single panel.

Figure 5 clearly indicates that the bonding layer ads significant vibration damping. However, the damping difference between the two cases cannot solely be attributed to the adhesive layer because of the sample size difference of both experiments. Since the loss factor expresses the ratio of dissipated vibration energy to the amount of vibration induced energy the amount of damping due to the adhesive layer can be estimated. Table 7 compares the estimated loss factors of the adhesive layer (from structural modal analysis) with the loss factors determined from dynamical material analysis.

f (Hz)	60	94	146	191	226	323	557
$\eta$ (/)	0,0904	0,0468	0,02	0,0106	0,011	0,0244	0,082

**Table 7:** Obtained values for the loss factors of the adhesive bonding layer.

These results match the loss factor values of table 3. Differences arise from the varying loading conditions applied in the different experiments and the geometrical differences of the test structures.

## 6 Conclusions and research prospects

This paper describes the process of estimating the main elastic properties of the adhesive layer in a simple bonded joint of two rectangular thermoplastic honeycomb sandwich panels. This is feasible by means of a mixed numerical-experimental modal analysis approach. However, the estimation of damping contribution of the adhesive layer is not straightforward in case the bonded structures are characterized by high damping themselves.



Current research focusses are twofold. Focus is laid on the adequate estimation of elastic adhesive properties in more complex geometrical bonded joints. This includes a more statistical approach including a study of the variability incorporated by environmental conditions and bonded zone geometry.

Research efforts also aim at a mixed numerical-experimental approach for an optimal prediction of an adhesively bonded joint's lifetime.

## Acknowledgements

This research work has been made possible by the efforts of Bjorn Marchand and Pol Coudeville. Much of the experimental and numerical work is on their behalf.

## References

- [1] R.D. Adams, W.C. Wake, *Structural adhesive joints in engineering*,
- [2] C. W. Bert, *Material Damping: An Introductory Review of Mathematical Models Measures and Experimental Techniques*, *Journal of Sound and Vibration*, Vol. 29, 1973.
- [3] C.T. Sun, Y.P. Lu, *Vibration Damping of Structural Elements*, Prentice-Hall, NJ, 1995.
- [4] B.J. Lazan, *Damping of Materials and Members in Structural Mechanics*, Pergamon Press, New York, NY, 1968.
- [5] L. da Silva, A. Öchsner, *Modeling of adhesively bonded joints*,
- [6] L.D.R. Grant, R.D. Adams, L. da Silva, *Experimental and numerical analysis of single-lap joints for the automotive industry*, *International journal of adhesion and adhesives*, vol. 29, iss. 4, June 2009, 405-413.
- [7] A. Sheppard, D. Kelly, L. Tong, *A damage zone model for failure analysis of adhesively bonded joints*, *International Journal of adhesion and adhesives*, vol. 18, iss. 16, dec. 1998, 385-400.
- [8] K. S. Madhusudhana, R. Narasimhan, *Experimental and numerical investigations of mixed mode crack growth resistance of a ductile adhesive joint*, *International journal of engineering fracture mechanics*, vol. 69, iss. 7, may 2002, 865-883.
- [9] W.C. De Goei, M. J. L. van Tooren, A. Beukers, *Composite adhesive joints under cyclic loading*, *International journal of materials and design*, vol. 20, iss. 5, October 1999, 213-223.
- [10] A. J. Kin Loch, S.J. Shaw, *The fracture resistance of a toughened epoxy adhesive*, *international journal of adhesion*, vol. 12, iss. 1, 1981, 59-77.
- [11] U. Edlung, A. Klarbring, *A coupled elastic-plastic damage model for rubber-modified epoxy adhesives*, *International journal of solids and structures*, vol. 30, iss. 19, 1993, 2693-2708.
- [12] L.D.R. Grant, R.D. Adams, L. da Silva, *Effect of the temperature on the strength of adhesively bonded single lap and T joints for the automotive industry*, *International journal of adhesion and adhesives*, vol. 29, iss. 5, July 2009, 535-542.

- [13] L. da Silva, P. das Neves, R.D. Adams, Wang A., Spelt J.K., *Analytical models of adhesively bonded joints—Part II: Comparative study*, International journal of adhesion and adhesives, vol. 29, iss. 13, 2009, 331-341.
- [14] T. Lauwagie, *Vibration-based methods for the identification of the elastic properties of layered materials*, Phd thesis, KU Leuven, D/2005/7515/80, 2005.
- [15] G.I. Schuëller, H.J. Pradlwarter, *Uncertain linear systems in dynamics: Retrospective and recent developments by stochastic approaches*, Engineering Structures, Volume 31, Issue 11, November 2009, Pages 2507-2517.
- [16] A. Schenk, G.I. Schuëller, *Uncertainty assessment of large finite element systems*, Springer, Innsbruck, 2005.
- [17] D. Moens, D. Vandepitte, *A survey of non-probabilistic uncertainty treatment in finite element analysis*, Computer Methods in Applied Mechanics and Engineering, Vol. 194, Nos.12-16, Pages 1527-1555, 2005.
- [18] F. Mortensen, O.T. Thomsen, *Analysis of adhesive bonded joints: a unified approach*, International journal of composites science and technology, vol. 62, iss. 7-8, June 2002, 1011-1031.
- [19] G. Richardson, A. D. Crocombe, P.A. Smith, *A comparison of two- and three- dimensional finite element analyses of adhesive joints*, International journal of adhesion and adhesives, vol. 13, iss. 3, July 1993, 193-200.
- [20] W. Renton, J. Vinson, *Analysis of adhesively bonded joints between panels of composite materials*, International journal of applied mechanics, 44(1), 1977, 101-106.
- [21] F. Ascione, *Mechanical behaviour of FRP adhesive joints: A theoretical model*, Composites part B: Engineering, vol. 40, iss. 2, 2009, 116-124.
- [22] S. Debruyne, *Estimation of skin and core stiffness properties of honeycomb sandwich panels*, Phd thesis, KU Leuven, D/ 2013/7515/64, 2013.
- [23] E. Ungar, E. Kerwin, *Loss Factors of Viscoelastic Systems in Terms of Energy Concepts*, Journal of the Acoustical Society of America. 34(7), 1962.
- [24] ISO Standard 6721-3, *Plastics - Determination of Dynamic Mechanical Properties, Part 3: Flexural Vibration-Resonance-Curve Method*, 1994.